International Journal of Engineering and Advanced Research Technology (IJEART)
ISSN: 2454-9290, Volume-3, Issue-4, April 2017

Research on time-delay vibration damping control of three degrees half-vehicle suspension system in idle condition

Qingchang Wang, Chuanbo Ren, Lei Zhang

***Abstract*— A 3DOF(three-degree of freedom) half-vehicle suspension model with time-delay feedback control is established to study vehicle vibration performance in idle condition. When running in idle condition, the vehicles don't have speeds and road excitation, so engine vertical self-vibration is the only excitation source. Utilizing Routh-Hurwitz criterion and a new frequency scanning method to analyze the stability of the suspension system. Introduced by the damping mechanism of time-delay dynamic vibration absorber and the system amplitude frequency characteristic, establishing the unified-object function based on the body displacement and pitch movement. By using Matlab Optimization Toolbox, designed the best time-delay feedback coefficient and time-delay value of suspension system, and the simulation is conducted in both time and frequency domain. Simulation results show the RMS(root-mean square) of vehicle body displacement and pitching motion with time-delay feedback control is respectively reduced 65.1% and 61.1% under the engine simple harmonic excitation. In conclusion, the vibration control with time-delay can effectively improve engine vibration isolation and vehicle working performance in idle condition, which provides a theoretical basis and reference for simulation analysis and optimal design of the vehicle vibration system in idle condition.**

***Index Terms*— idle condition; suspension system; time-delay damping; stability; Matlab Optimization Toolbox**

I. INTRODUCTION

The engine is one of the most important vibration sources of the vehicle. For four-stroke engines, crankshaft speed in power stroke is higher than the other. The rotation speed in cyclical changes cause rigid body vibration and elastic vibration of the power assembly mounting system, so non-uniform crankshaft speed is the main reason for the engine vibration.[1] Furthermore, abnormal combustion of gasoline, exhaust fan, parts impact, flywheel and other eccentric rotation also lead to engine vibration. The engine vibration through the suspension system to the body causes a series of resonance, resulting in vehicle structural complex fatigue and noise problems. [2] Therefore, the main design goal of the engine suspension is to reduce the transmission of the engine vibration to the body, reduce the vibration amplitude of the car powertrain to avoid interference, and optimize the ride comfort in idle condition. But the technology development trend of vehicle powertrain system is put forward higher requirement, which is mainly manifested in lightweight of automobile causes the body more sensitive to vibration, and the reduction of motor cylinder number makes deterioration in the balance of powertrain. In addition, FF(Front-engine Front-drive) layout configuration also has adverse effects on smoothness. Under the idling conditions, the engine itself becomes the main vibration sources, its frequency is generally above 20Hz. At this time the transmission device requires a smaller stiffness and damping in order to reduce the vibration to the body.

Suspension system, as an important part of the car chassis, is playing a key role in vehicle ride comfort. Many scholars have done a lot of theoretical and experimental research on vehicle suspension since proposing the theory of semi-active and active suspension. With the continuous improvement of the suspension system in the control field, the precision of the control system is continuously improved. However, the implementation of active control strategy can't be separated from signal collecting and processing in the actual project, and in order to achieve better suspension performance, more efficient high-speed signal processing is necessarily. Meanwhile, time lag between signal acquisition and execution must be considered.[3]-[4] Therefore, it is of great theoretical and practical value to design the active suspension with time-delay, ensure the stability of the control system and improve the vibration performance of the suspension.

In this paper, the time-delay damping technique is introduced for the 3-DOF（three-degree of freedom）half-car suspension model in idle condition to control auto body principal vibration. Because vehicles don't have the running speed and road excitation in idle condition, engine self-vibration is the only excitation sources. When the engine is symmetrically distributed along the central axis, the vibration characteristic of suspension is exactly symmetrical. Therefore, the three degree-of-freedom half-car model can reasonably simulate the vibration characteristic under idling conditions. The vehicle body displacement and pitching motion are used as the objective function to design and optimize in the paper.[5] The stability of the suspension system is analyzed by using Matlab Optimization Toolbox, getting the optimal feedback gain and time delay. And the simulation of suspension model is conducted in both time and frequency domain.

II. MECHANICAL MODEL

The car is a complex multi-degree-of-freedom vibration system with a lot of uncertainty, time-variation and nonlinear. In order to conveniently analyze the influence of time-delay control on suspension system, the vehicle is reasonably simplified as a linear model. Engine vibration is the main

factor on affecting ride comfort under idling condition. When the engine is symmetrically distributed along the central axis, the vibration performance of the suspension is exactly symmetrical. Therefore, the 3-DOF half-car model can simulate the vibration characteristics under idling conditions reasonably. A vehicle model is simplified in figure 1, including the displacement and pitch movement in the body with two degrees of freedom and the engine displacement with one degree of freedom.[6]-[7]

Using Lagrange equation and Newton's second law, delay-time differential equations of the suspension system is established as shown in Eq.(1)

$$\begin{cases} m_e\ddot{x}_e + k_e\left(x_e - x_{sf}\right) + c_e\left(\dot{x}_e - \dot{x}_{sf}\right) - f_d + f = 0 \\ m_c\ddot{x}_c + k_{sf}x_{sf} + c_{sf}\dot{x}_{sf} + k_{sr}x_{sr} + c_{sr}\dot{x}_{sr} - k_e\left(x_e - x_{sf}\right) - c_e\left(\dot{x}_e - \dot{x}_{sf}\right) - f = 0 \\ I_c\ddot{\varphi} - L_f\left(k_{sf}x_{sf} + c_{sf}\dot{x}_{sf}\right) + L_r\left(k_{sr}x_{sr} + c_{sr}\dot{x}_{sr}\right) - L_e\left[k_e\left(x_e - x_{sf}\right) + c_e\left(\dot{x}_e - \dot{x}_{sf}\right) + f\right] = 0 \end{cases} \quad 1)$$

There, $f_d = f_n\cos(\omega_0 t)$, $f = gx_e\left(t-\tau\right)$

As the body pitching angle is small, it can be approximated: $\begin{cases} x_{sf} = x_c - L_f\varphi \\ x_{sr} = x_c + L_r\varphi \end{cases}$

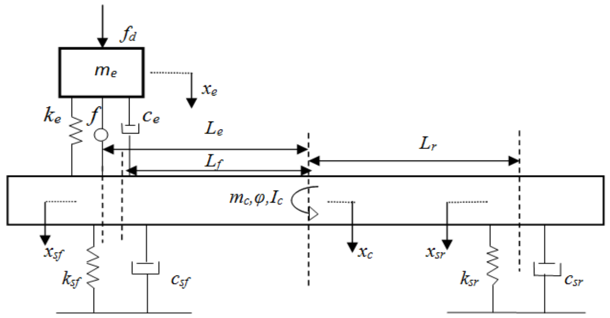

Fig.1 3-Degree of freedom semi-car model in idle condition

where, L_f、L_r is the distance from vehicle body centroid to the front and rear axles respectively, as is shown in table 1; L_e is the distance from vehicle body centroid to the engine mounting system; m_c 、I_c representing the body quality and moment of inertia respectively; m_e is the quality of engine; k_{sf}, c_{sf}, respectively, is the stiffness coefficient and damping coefficient of front suspension; k_{sr}, c_{sr}, respectively, is the stiffness coefficient and damping coefficient of rear suspension; k_e, c_e, respectively, is the stiffness coefficient and damping coefficient of rear suspension engine mounting system; g is the gain coefficient of displacement feedback; τ is delay value.

According to the characteristics of ordinary gasoline engines cars, we can realize that the speed of the ordinary inline four-stroke, four-cylinder engines is generally not more than 800r/min in idle condition. At this point, the engine vibration frequency is 25Hz, which is two times as much as the frequency of the crankshaft. The vertical force of engine mounting system is generally not more than 300N under the limited of powertrain layout.

Tab.1 Model parameters of vehicle suspension

L_f /m	L_r /m	L_e /m	m_c /kg	I_c /kg·m^2	m_e /kg
1.3	1.5	1.4	690	1222	60
k_e /N·m^{-1}	k_{sf} /N·m^{-1}	k_{sr} /N·m^{-1}	c_e /N·s·m^{-1}	c_{sf} /N·s·m^{-1}	c_{sr} /N·s·m^{-1}
150000	17000	22000	1100	1500	1500

III. STABILITY ANALYSIS OF TIME-DELAY SYSTEM

The Laplace transform is applied to the system differential equations:

$$\det(\mathbf{A}_{33}) = \det\begin{bmatrix} a_{11} & a_{12} & a_{13} \\ a_{21} & a_{22} & a_{23} \\ a_{31} & a_{32} & a_{33} \end{bmatrix} = 0 \quad (2)$$

There,

$a_{11} = m_e s^2 + sc_e + k_e + ge^{-s\tau}$, $a_{12} = -(k_e + sc_e)$,

$a_{13} = L_f k_e + sL_f c_e$,

$a_{21} = -(sc_e + k_e + ge^{-s\tau})$,

$a_{22} = m_c s^2 + (k_{sf} + k_{sr} + k_e) + s(c_{sf} + c_{sr} + c_e)$,

$a_{23} = (L_r k_{sr} - L_{sf} k_f - L_f k_e) + s(L_r c_{sr} - L_f c_{sf} - L_f c_e)$,

$a_{31} = L_e(sc_e + k_e + ge^{-s\tau})$,

$a_{32} = (L_r k_{sr} - L_f k_{sf} - L_e k_e) + s(L_r c_{sr} - L_f c_{sf} - L_e c_e)$,

$a_{33} = I_c s^2 + L_f(L_f k_f + L_e k_e) + L_r^2 k_r$,
$+ s[L_f(L_f c_f + L_e c_e) + L_r^2 c_r]$

The characteristic equation (2) can be written in a polynomial:

$$CE(s,\tau) = a_{10}s^{10} + a_9 s^9 + a_8 s^8 + a_7 s^7 + a_6 s^6 + a_5 s^5 + a_4 s^4 + a_3 s^3 + a_2 s^2 + a_1 s + a_0 + be^{-\tau s} = 0$$

Where, $a_{10} \ldots a_0$, b are the characteristic equation coefficients. Because the numerical values are large, only reflected in the calculation process by Maple language, this article is no longer stated.

①When the time-delay τ=0, the system characteristic equation is shown in Eq.(3),

$$a_{10}s^{10} + a_9 s^9 + a_8 s^8 + a_7 s^7 + a_6 s^6 + a_5 s^5 + a_4 s^4 + a_3 s^3 + a_2 s^2 + a_1 s + a_0 = 0 \quad (3)$$

Using the Routh-Hurwitz criterion, the necessary and sufficient conditions for the system stability are obtained.

When the coefficient symbols of the equation(3) are all the same, and the first column of the Rolls Series are all positive numbers, the feedback gain is:

g>-150000N/m

②When the time-delay value τ >0, $s = \omega i$ will be substituted into equation (2), and using Euler's formula $e^{-i\omega\tau} = \cos(\omega\tau) - i\sin(\omega\tau)$ to substitution, the separation of real and imaginary part is available in Eq.(4)

$$\begin{cases} \mathrm{Re}(\det(\mathbf{A}_{55})) = 0 \\ \mathrm{Im}(\det(\mathbf{A}_{55})) = 0 \end{cases} \quad (4)$$

The functional equation (5) for g, ω can be solved.

$$\begin{cases} \cos(\omega\tau) = F_1(g,\omega) \\ \sin(\omega\tau) = F_2(g,\omega) \end{cases} \quad (5)$$

According to the triangular relation $\sin^2(\omega\tau) + \cos^2(\omega\tau) = 1$, we can get the equation $g = G(\omega)$, it's g only about ω; and bring it into equation (5) to obtain equation $\tau = \mathrm{T}(\omega)$. The parametric equations

$\begin{cases} g = G(\omega) \\ \tau = T(\omega) \end{cases}$ give the complete stable region with different feedback gain g and delay τ in figure 2, when ω is within the critical frequency.[8]-[11]

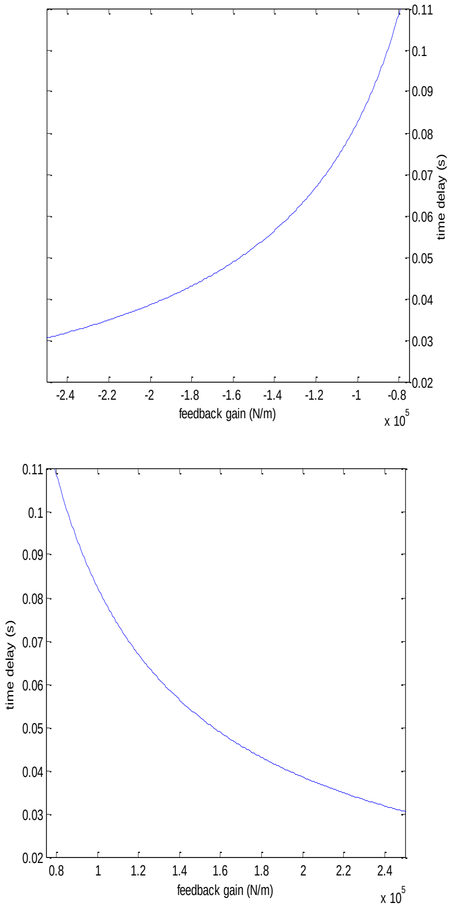

Fig.2 Critical stability curve

It can be seen from the figure that due to the feedback gain occurs in square form, the stable region is symmetrical about the time-delay axis. And when g is less than a certain value, the system can be stable regardless of the τ value.

IV. Control Parameter Optimization

If the excitation force has a Fourier transform, $f_d(t) = F_d(\omega)$, then the system differential equation (1) can be written as Fourier transform, as shown in Eq.(6).

$$A_{55}\begin{bmatrix} X_e(\omega) \\ X_c(\omega) \\ \varphi(\omega) \end{bmatrix} = \begin{bmatrix} F_d(\omega) \\ 0 \\ 0 \end{bmatrix} \qquad (6)$$

The response of the vertical displacement and pitch movement in the body to the input excitation can be obtained by Eq.(5)

$$|H_{Xc}(\omega)| = \left|\frac{X_c(\omega)}{F_d(\omega)}\right| = \left|\frac{a_{23}a_{31} - a_{33}a_{21}}{\det(A_{33})}\right|$$

$$|H_{\varphi}(\omega)| = \left|\frac{\varphi(\omega)}{F_d(\omega)}\right| = \left|\frac{a_{32}a_{21} - a_{22}a_{31}}{\det(A_{33})}\right|$$

In order to consider the relative importance of each sub-objective function in the whole multi-objective function, we introduce the weight coefficient c_1 and c_2. And the unified-object function is established, as shown in Eq.(7)

$$Min\ J(\omega) = c_1|H_{Xc}(\omega)| + c_2|H_{\varphi}(\omega)| \qquad (7)$$

Here, c_1=0.3, c_2=0.7.

According to the physical background, the delay τ can only take a positive real number and less than the critical delay, and the gain g is also not too large to get out of the actual engineering. The basic parameters of the system shown in the paper are taken into the objective function, the optimal parameters g=-155000*N/m* and τ=0.02*s* are obtained by Matlab Optimization Toolbox.[12]

V. Vibration Response Analyses

In order to test the influence of delay time in the vibration system, the time domain and frequency domain response are given in the paper.

A. *Frequency domain characteristics*

Bring optimal parameters g=-155000*N/m*，τ=0.02*s* into the amplitude-frequency characteristic function of vibration response, and getting the amplitude-versus-frequency curve as shown as figure 3.

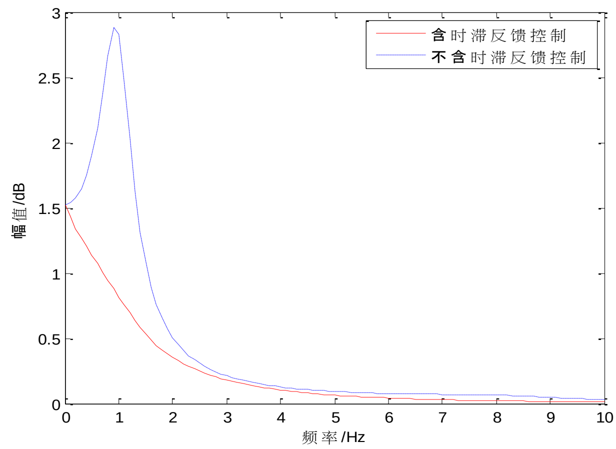

Fig.3 feedback frequency domain

It can be seen from the figures that the time-delay feedback control has a great influence on the objective function. Compared with the no time-delay control, the amplitude of the objective function has different degrees of decrease in the frequency range of the change. In the frequency range of 0~ 2Hz, the amplitude of the objective function with time-delay are significantly reduced compared with the passive feedback system. When the frequency is higher than 2Hz, compared with no time-delay control system, the amplitude of the objective function is not significant, and their amplitudes are both small, so with or without feedback on the whole system has little effect in this range of frequency. It can be concluded that the active system with time-delay feedback control can reduce the vibration characteristics of the suspension and optimize the smooth performance of the vehicle compared with the passive system. If we reasonably control the size of the time-delay, making it reasonable matching with the feedback system, the system vibration can be significantly reduced. Indicating that the time-delay feedback than the traditional passive feedback has great advantages in a certain frequency range.

B. Time domain simulation

Compared passive feedback system without time-delay and active feedback system with time-delay, analyzing time-domain simulation of the system's vibration response under the action of simple harmonic excitation ($f_d=300cos(50t)$), as shown in figures 4-8.

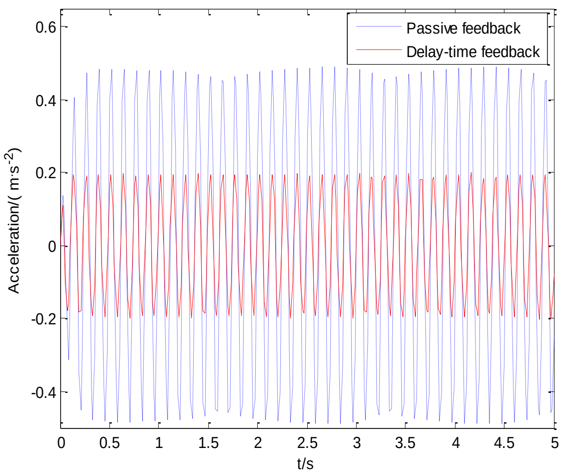

Fig.4 Body centroid acceleration

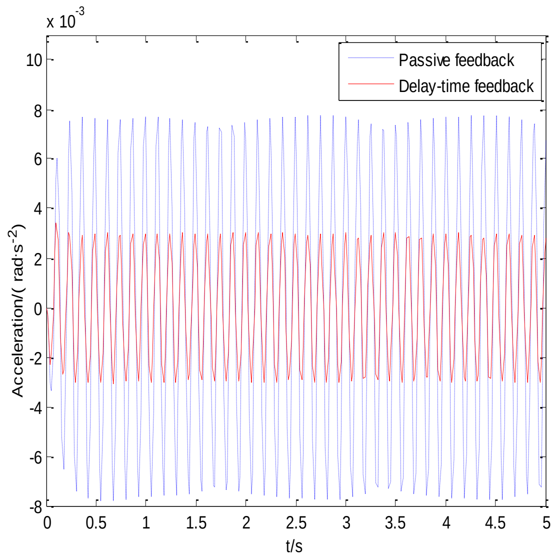

Fig.5 Body pitching acceleration

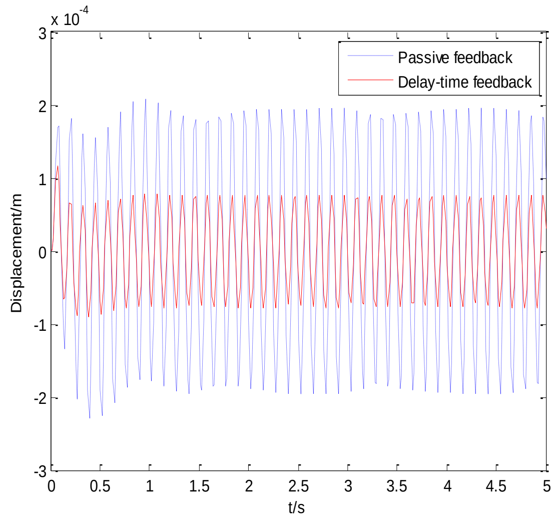

Fig.6 Body centroid displacement

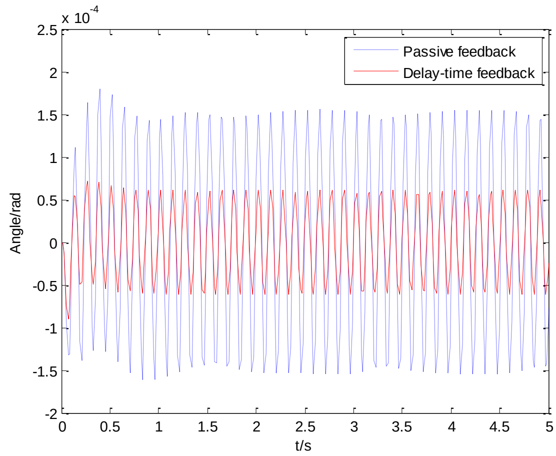

Fig.7 Body pitching angle

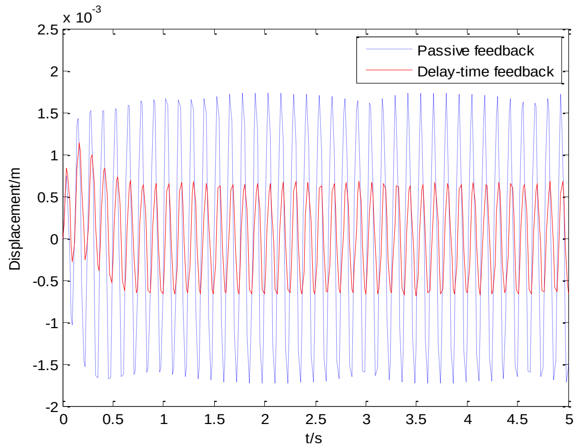

Fig.8 Engine displacement

The simulation results show that the time-delay feedback control has a great advantage over the passive feedback control, and its corresponding smooth performance indexes are reduced to different degrees. Compared with the passive feedback, the active suspension with delay feedback control significantly reduces the RMS values of the vehicle body centroid and pitch angle acceleration, and the vibration reduction efficiency is 64.0% and 63.4%, as shown in Table 2. The RMS values of centroid and pitch displacement of the vehicle are reduced by 65.1% and 61.1% respectively. The RMS of the engine vertical displacement decreased from 1.62*mm* to 0.70*mm* and the vibration reduction efficiency was 56.8%. It can be concluded that under the idle condition, only the engine itself vibration as the external excitation, the active suspension with time-delay control can be a good way to reduce the vehicle's vibration response and optimize the performance index.

Tab.2 The RMS value of smoothness index

Indicators/Unit	*RMS*		*Optimization proportion(%)*
	Passive feedback	*Time-delay feedback*	
Body centroid acceleration / $m{\cdot}s^{-2}$	0.478	0.172	64.0
Body pitch acceleration / $rad{\cdot}s^{-2}$	$7.79*10^{-3}$	$2.85*10^{-3}$	63.4
Body centroid displacement /*m*	$1.95*10^{-3}$	$0.68*10^{-3}$	65.1
Body pitching angle/*rad*	$1.49*10^{-4}$	$0.58*10^{-4}$	61.1
Engine displacement/*m*	$1.62*10^{-3}$	$0.70*10^{-3}$	56.8

VI. Conclusion

In this paper, the time delay feedback is introduced for the three-degree of freedom half-car model, and studying the vehicle vibration that affected by the engine vertical vibration under the idle condition. The suspension model with the delay feedback control is optimized and simulated numerically. Getting the following conclusions:

Firstly, for the 3-DOF suspension system under idle conditions, the feedback control based on the vertical displacement of the engine is introduced by using time-delay dynamic vibration absorber theory. The dynamic model with delay feedback is established, and the stability region of the suspension system is obtained.

Secondly, the vertical displacement and pitch motion of the vehicle body are taken as the objective function in the vertical harmonic excitation of the engine, and the time delay parameter is optimized by MATLAB toolbox. The simulation results show that the vertical centroid and pitch angular displacement are reduced by 65.1% and 61.1% respectively, and the vibration is obviously improved. Body centroid acceleration and engine displacement are also significantly reduced. The results show that the delay control can effectively improve the vibration reduction effect and improve the ride comfort and smoothness of the vehicle.

Thirdly, considering the vibration control with time-delay can effectively optimize the engine suspension system, enhance the vibration isolation performance of the engine, improve the performance of the vehicle under idling conditions, which provides a theoretical basis and design reference for the simulation analysis and optimization design of the vehicle vibration system under idle condition, and has a certain application value.

ACKNOWLEDGMENT

This subject is supported by Natural Science Foundation of China (51275280).

REFERENCES

[1] Dong L. “Study on vibration testing and vibration isolation optimization of gasoline engine”. JiLin University, pp. 4-15, 2012.
[2] Chen W J. “Theoretical and experimental study on abnormal vibrations of idling engines”. Forestry machinery and woodworking equipment, pp.2-4, 2008.
[3] Lei J, Tang G Y. “Optimal vibration control for active suspension systems with actuator and sensor delays”. Proceedings of IEEE International Conference on system, pp.2830-2833, 2008.
[4] Zhang W F, Weng J S, Hu H Y. “Effect of time delay on active vehicle suspensions equipped with “Sky-Hook” damper”. Journal of Vibration Engineering, pp.489-490, 1994.
[5] Jalili N, Olgac N. “Multiple delayed resonator vibration absorbers for multi degree of freedom mechanical structures”. Journal of Sound and Vibration, vol.4, no.223, pp.568-585, 1999.
[6] Wang Z H, Hu H Y. “Stability switches of time-delay dynamics systems with unknown parameters”. Journal of Sound and Vibration, vol.4, no.233, pp.219-231, 2000.
[7] Tang G Y. “Feedforward and feedback optimal control for linear systems with sinusoidal disturbance”. High Technology Letters, vol.4, no.7, pp.16-18, 2001.
[8] Renzulli ME, Roy RG, Olgac N. “Robust control of delayed resonator vibration absorber”. IEEE Transactions on Control Systems Technology, vol.6, no.7, pp.688-691, 1999.
[9] Chu S Y, Song T T, Lin C C, et al. “Time-delay effect and compensation on direct output feedback controlled mass damper systems”. Earthquake Engineering & Structural Dynamics, vol.1, no.31, pp.125-133, 2006.
[10] Zhao Y Y, Xu J. “Delayed resonator and its effects on vibrations in primary system”. Journal of Vibration Engineering, vol.4, no.19, pp.549-553,2006.
[11] Gao Q. “Combination of sign inverting and delay scheduling control concepts for multiple-delay dynamics”. Systems and Control Letters, vol.4, no.77, pp.56-60, 2015.
[12] Deng W, Li C, Lü J. “Stability analysis of linear fractional differential system with multiple time delays”. Nonlinear Dynamics, vol.4, no.48, pp.409-416, 2007.

Qingchaung Wang. He was born on May 17, 1992 in Shandong province. He is a second-year graduate student at the Shandong University of Technology. His research direction is the vehicle vibration control with time-delay.

Chuanbo Ren , He was born on 1964 in Shandong province, China. He is a professor and PhD Tutor at the Shandong University of Technology. His main research direction is he engineering mechanics optimization and vehicle system dynamics.

Lei zhang. He was born on 06th December 1992 in Shandong province, China. He is a postgraduate student at the Shandong University of Technology, and major in vehicle engineering. His research direction is the vehicle system dynamics